

THE NECESSITY OF AN ADAPTIVE VEHICLE STRUCTURE TO OPTIMIZE DECELERATION PULSES FOR DIFFERENT CRASH VELOCITIES

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ABSTRACT

To minimize injury to the occupants, the frontal vehicle structure must absorb much more energy in the first deformation phase in case of a high-speed collision. Depending on the crash situation an intelligent system must regulate the structure stiffness yielding additional energy absorption by means of friction. Concept ideas are mentioned to achieve different crash pulses at different crash velocities within the available deformation length.

An independent search for optimal deceleration pulses at several crash velocities is necessary, because the usually found structure-based pulses are not obviously the optimal pulses for minimal injury to the occupants. Therefore, in this paper the more interesting case of the reverse question is answered: which crash pulse gives the lowest injury levels with an already optimized restraint system, instead of finding the optimized restraint system for a given crash pulse. For this research, a method is described in which a numeric model of an interior and a FEM dummy has been used to find the levels of the injury criteria. To compare the results of different crash pulses, an overall severity index has been used. From a described research an optimal pulse has been found after several considered pulse variations at a crash speed of 56 km/h. This pulse, used as example, gives as it seems much lower injuries. During the first 18 cm deformation length the deceleration level must be high, then a low deceleration interval is required, and at the end (dummy is restrained by belt and airbag) the deceleration must be high again. Also for other crash velocities, pulses are mentioned with adapted pulse characteristics for optimal results.

The only way to generate an optimal crash pulse at different collision speeds is variable structure stiffness. After detection of the crash velocity, the optimal stiffness of the front structure should be realized. Solutions are presented based on controllable energy absorption by additional friction or based on controllable hydraulic flow restriction. With this new design, an optimal vehicle deceleration curve is possible for each velocity over the entire

frontal collision spectrum, yielding the lowest levels of the occupant injury criteria, also in case of compatibility problems.

INTRODUCTION

The improved frontal crashworthiness of cars necessitates totally new design concepts, which take into account that the majority of collisions occur with partial frontal overlap and under off-axis load directions. Realistic crash tests with partial overlap have shown that conventional longitudinal structures are not capable of absorbing all the energy in the car front without deforming the passenger compartment.

For improved frontal car safety it is necessary to design a structure that absorbs enough energy in each realistic crash situation. To protect the occupants, the passenger compartment should not be deformed and intrusion must be avoided too. To prevent excessive deceleration levels, the available deformation distance in front of the passenger compartment must be used completely for a predetermined crash velocity. This implies that in a given vehicle concept the structure must have a specific stiffness. Normally, the two main longitudinal members will absorb most of the crash energy with a progressive folding deformation of a steel column. The main problem is that in real car collisions these two longitudinal members often are not loaded in a synchronous fashion. The majority of collisions occur with partial frontal overlap, in which only one longitudinal is loaded. A design conflict is that the same amount of energy must be absorbed either with a single or with both longitudinals. These problem can not be solved by just increasing the stiffness of the longitudinals in such a way that each longitudinal is capable of absorbing all of the energy, see the following reasons. To absorb enough energy, a stiff longitudinal is needed for the offset crash in which normally only one longitudinal is loaded. The same longitudinal must be more supple in case of a full overlap crash, since both longitudinals must not exceed the desired deceleration level (Witteman 1993). Another issue is the crash velocity. To absorb all the kinetic energy, which is proportional with the square of the velocity, the deformable structure length

must have a specific stiffness. This stiffness results in an average mean force, which multiplied with the deformation shortening gives the absorbed energy. For an acceptable injury level of the occupants, the total deceleration level must be as low as possible, using the maximum available deformation length without deforming the passenger compartment. This means that for example in a 64 km/h crash compared with a 32 km/h crash, a four times longer deformation distance is needed for the same deceleration level. Although the stiffness normally increases during the crash and at higher crash speeds there is made use of the stiff engine; the only way to generate an optimal crash pulse at different collision speeds is variable structure stiffness. After detection of the crash velocity, the optimal stiffness of the longitudinal member should be realized.

The objective of the research project presented here, was to design a concept structure that substitutes the conventional energy absorbing longitudinal members in a frontal vehicle structure and yields optimized deceleration pulses for different crash velocities and overlap percentages. To this aim the structure must have a stiffness that can be varied in accordance to the specific crash situations.

The novel design presented in this paper can cope with the following three crashworthiness problems:

1. In the case of a full overlap crash (both longitudinals and engine involved) as in the case of an offset crash (at 40 per cent overlap only one longitudinal directly involved) a similar amount of energy must be absorbed by the front structure.
2. With a not much longer deformation length, much more energy must be absorbed at high crash velocities (resulting in less fatal injuries) and a lower injury level must be obtained at lower crash velocities.
3. A deceleration pulse must be obtained which is optimal (lowest injury level) for the concerning collision speed and the chosen dummy restraint parameters.

METHOD FOR OPTIMIZING THE DECELERATION PULSE

The aim of this research is to minimize the injury level of the occupant in several frontal collisions. Therefore, it must be clear which parameters influence the injury level. If an undeformable passenger compartment and no intrusion of vehicle parts like steering wheel, dashboard and pedals are assumed, the injury level is only influenced by means

of g forces of the deceleration pulse generated by the vehicle front. To be sure that the injury level is the lowest possible, a numerical model is necessary to calculate the expected injury level by variation of the deceleration pulse. If the optimal deceleration pulse for a specific crash velocity is known, the structure must be designed to generate such a desired crash behavior.

With an ideal not deforming passenger compartment, it is acceptable to use an uncoupled model of the dummy and the frontal deforming structure. A common method is, to predefine a deceleration pulse as input on the passenger cage. A full frontal coupled model has a longer calculation time, also because the dummy movement has a longer crash duration time while the frontal structure is already fully deformed. The usual real time interactions between the occupant and the vehicle structure during a crash (Khalil 1995), which influence the vehicle deceleration a little (Seiffert 1992) by means of the restrained dummy mass, can be compensated in the input pulse. Of course for exactly determining the deceleration pulse of a vehicle structure (not for determining the pulse that causes the lowest injury level) the dummy masses must be added to the vehicle model with restraint characteristics. Of course in case of a side impact an uncoupled method is not allowed, the dummy mass and its close position to the door have a not negligible influence on the deformation behavior of the relatively low mass of the side structure (Landheer 1996).

With the aid of an interior model, variations of the deceleration pulse can be compared on basis of a calculated injury level and an optimal pulse can be obtained for several crash velocities. Structural design specifications are presented to realize such an optimal pulse and conceptual design ideas will be proposed which fulfil these desired deceleration levels for different crash velocities.

To compare the injury severity for different vehicle collisions, some sort of index or formula is needed. The regulations for vehicle crashes only prescribe a maximum value not to be exceeded for several different injury criteria. Because the proposed vehicle model has no intrusion, only the injury criteria as mentioned in Table 1 with their limiting values (Levine 1994, Mertz 1993, Seiffert 1997) are useful. An overall severity index can be a specific weighted combination of these injury criteria, and which takes also into account the relative importance of individual changes of these injury criteria (Bakker 1997). The relative importance to an overall severity index can be expressed by a weight factor (Viano 1990). For an

extended description of an overall severity index see a separate research of Witteman (1999b).

Table 1.
Relevant injury criteria with their by legislation limited values

Injury Criterion	HIC	CHEST-G	CHEST-D	FEMUR-F	NECK-M
Limit value	1000	60 g	50 mm	10000 N	189 Nm
Weight factor	8	2	2	1	2

The Head Injury Criterion (HIC) is calculated on a specific time interval around a deceleration peak of the dummy head to reach a maximum value as shown in next formula, where t_1 and t_2 are the start and end time of the considered deceleration interval in seconds and $a(t)$ is the head deceleration in g as function of time:

$$HIC = \max \left\{ \frac{1}{(t_2 - t_1)} \cdot \int_{t_1}^{t_2} a(t) dt \right\}^{2.5} \cdot (t_2 - t_1)$$

CHEST-G is the peak deceleration in g of the dummy chest.

CHEST-D is the peak compression of the dummy chest, mostly a result of the belt force.

FEMUR-F is the maximal longitudinal force in the upper leg caused by the dashboard.

NECK-M is the flexion bending moment of the dummy neck by forward head rotation.

To simulate the movement behavior of an occupant and to measure the forces working on the body, use can be made of a modern deformable frontal finite element dummy Hybrid III (ESI 1996). This dummy consists of rigid body elements and a full deformable thorax and pelvis and is developed by the safety group of ESI SA (Rungis, France) in collaboration with the Biomechanics Department of Wayne State University (Detroit, USA). This numerical dummy simulates the crash dummy of a full-scale frontal crash. In literature a good correlation is reported between computed accelerations of the basic rigid body dummy and measured accelerations in a sled test (Ni 1991). Also the new dummy with deformable chest and pelvis, shows good correlation's with low and high speed pendulum impact tests and with a sled test (ESI 1996, Schlosser 1995).

The dummy model must be positioned inside a realistic vehicle interior model (Bosch 1993, Relou 1995, Seiffert 1989, Wijntuin 1995). To this aim, a seat, a dashboard with steering wheel, a floor plane and a restraint system must be modeled, see Figure 1. The restraint system consists of a belt with automatic lock and a retractor, and a folded (as far as necessary) airbag (Hoffman 1989). The folding FEM airbag has a good interaction with a dummy and shows a good agreement with experimental data (Hoffmann 1990, Lasry 1991). The seat has a so-called anti submarining plate, which prevents forward moving of the dummy pelvis under the lap belt.

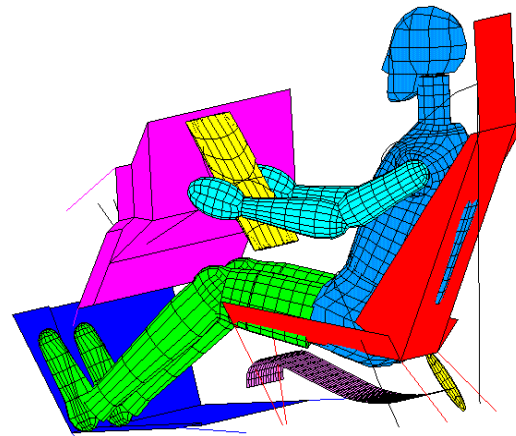


Figure 1. Dummy positioning within the interior with restraint systems.

A speed of 56 km/h against a rigid barrier is a realistic test speed because it gives a balance between acceptable injuries at lower speeds and minor fatalities at higher speeds.

RESULTS

Research has been carried out (Witteman 1999b) to find a deceleration pulse (the resulting deceleration-time signature on the vehicle passenger compartment generated when a collision occurs) with a minimal injury risk, based on the lowest value of the overall severity index, at a 56 km/h crash during 90 ms. This pulse determines the occupant loading and hence the injury risk for a passenger in a vehicle involved in an accident. In this research the reverse approach is used by answering the question which crash pulse gives the lowest injury levels with an already optimized restraint system, instead of finding the optimized restraint system for a given crash pulse. In Figure 2 the optimal pulse for 56 km/h is given with the corresponding velocity and deformation length curve against time, in which three deceleration levels can be

distinguished. These three phases can correspond successively with the crash initiation phase (sensor triggering), the airbag deployment phase and finally the occupant contact phase (Brantman 1991).

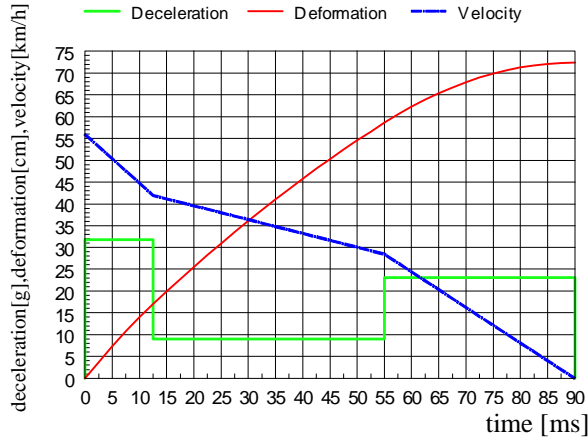


Figure 2. Optimal deceleration pulse and the velocity and deformation curve.

In Figure 3 the specific injury time plots of this new optimal pulse are given as an example. The injury values are plotted with a SAE180_5 filter with the times until 150 ms on the x-axis and on the y-axis for the HIC the deceleration in mm/ms^2 , for the CHEST-G also the deceleration in mm/ms^2 , for the CHEST-D the negative elongation (deflection) in mm of 7 bar elements perpendicular to the chest (where the CHEST-D is calculated as the average of this 7 distances), for the FEMUR-F the force in kN, and for the NECK-M the flexion moment in kNmm. Note the balanced course of the curves yielding the lowest injury values. For this optimal pulse the calculated injury values are given as indication in Table 2. The values are far below the limit values of Table 1.

Table 2.
Simulation results for the injury types of an optimized pulse at 56 km/h

Injury Criterion	HIC	CHEST-G	CHEST-D	FEMUR-F	NECK-M
Simulation value	251	31 g	21 mm	5066 N	29 Nm

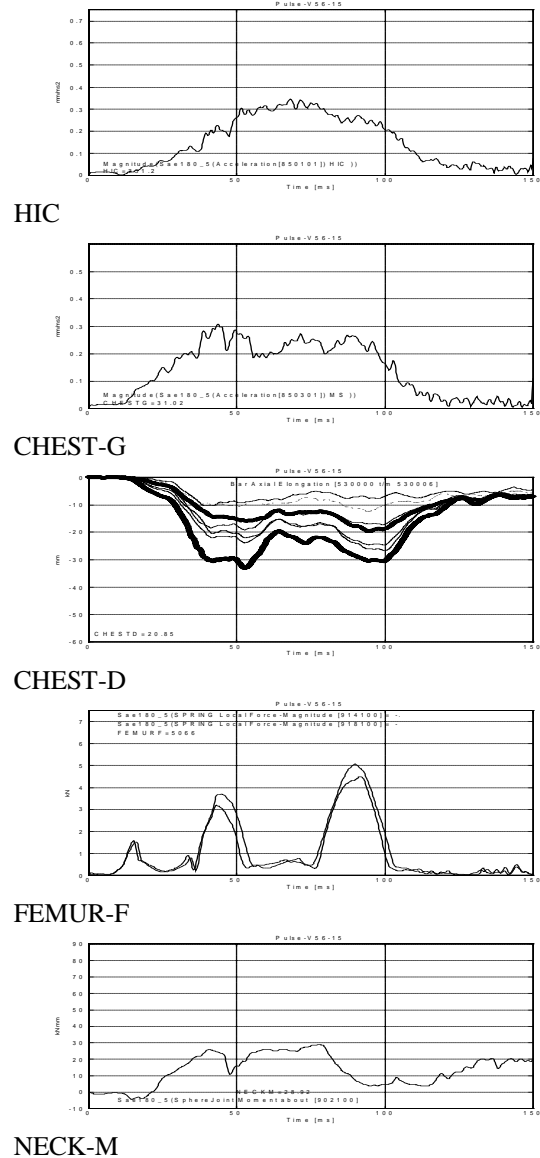


Figure 3. Injury values of an optimal pulse for 56 km/h.

Structural Design Specifications for Different Crash Velocities

Since more than 90 per cent of all frontal collisions occurs at a velocity lower than the prescribed crash velocity of 56 km/h (Witteman 1999a), also an optimal pulse for lower collision velocities is necessary to minimize the occupant injury level at that lower velocity. Although only 2 to 10 per cent (dependent of the overlap percentage) of all crashes takes place at a velocity higher than 56 km/h, also a pulse optimized at such a velocity is interesting because of the higher energy level yielding larger vehicle deformations and higher injury levels. If the

initial crash velocity is decreased to 32 km/h, this results in a decrease of crash energy of 67 per cent with respect to a crash speed of 56 km/h, so the vehicle might just be too stiff. An increase of the initial crash velocity to 64 km/h results in an increase of energy of 31 per cent, so the structure might be too supple. An optimal pulse for a speed of 64 km/h (total deformation length of 762 mm) is plotted in Figure 4 together with the already mentioned optimal pulse for 56 km/h (total deformation length of 724 mm) and with an optimal pulse for a collision with 32 km/h (total deformation length of 448 mm). These additional pulses are obtained with the same numerical research.

Before designing structural solutions to realize the desired deceleration pulses (see next sections), first the specifications for this design will be mentioned. For these specifications the optimal curves that were obtained for three different crash velocities as shown in Figure 4 will be used. In this figure the velocity decrease is plotted against deformation length instead of time, because it is more interesting to know on which length position in the car structural measures are necessary to realize a change in stiffness corresponding with the desired change in deceleration level. In Table 3 the time duration and deformation length of each deceleration interval are presented. As can be seen the difference in deformation length in the first interval of the 56 km/h and the 64 km/h collision is small. So for simplification the lengths of 170 mm and 188 mm could be joined together on 179 mm. At the end of the second interval the deformation length is already identical for both velocities, *vz.* 586 mm. These interval borders are visualized in Figure 4 as two vertical lines. For the 32 km/h collision there is no difference in deceleration between the first two intervals.

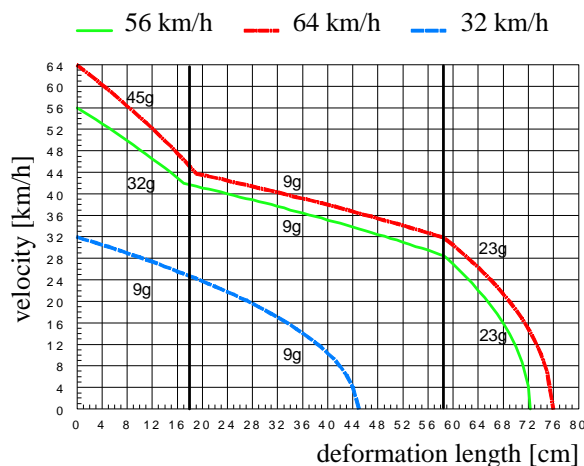


Figure 4. Three optimal decelerations curves in three phases (Wittelman 1999b).

The optimal pulse obtained for higher velocities has a higher deceleration level in the first interval and the levels of the middle and the third interval remain unchanged in comparison with an optimized pulse for 56 km/h.

The obtained optimal pulse for a collision with 32 km/h has a constant deceleration level of 9 g, the same level as the higher velocity pulses have during their middle interval.

From these observations it can be concluded with the considered numerical model, that for minimal injury for crash velocities starting at 32 km/h the vehicle structure needs a constant stiffness to decelerate 9 g during the first 586 mm. For higher velocities as 32 km/h the stiffness of the first 179 mm must be directly increased to decelerate up to 45 g for the highest velocity of 64 km/h. After 586 mm deformation has been reached the stiffness must be increased to decelerate to 23 g for all relevant velocities.

Table 3.

Deceleration parameters of 3 crash velocities (Wittelman 1999b)

Crash velocity	32 km/h	56 km/h	64 km/h
<i>Phase 1</i>			
Deceleration	9 g	32 g	45 g
Deformation length		170 mm	188 mm
Time duration		12.5 ms	12.5 ms
<i>Phase 2</i>			
Deceleration	9 g	9 g	9 g
Deformation length	total 448 mm	416 mm, total 586 mm	398 mm, total 586 mm
Time duration	100.7 ms	42.5 ms, total 55 ms	37.5 ms, total 50 ms
<i>Phase 3</i>			
Deceleration		23 g	23 g
Deformation length		138 mm, total 724 mm	176 mm, total 762 mm
Time duration		35 ms, total 90 ms	39.4 ms, total 89.4 ms

The Necessity of an Adaptive Structure

In the following sections, conceptual design ideas will be presented which can fulfil the specifications of different deceleration levels for an optimal deceleration pulse as given in Table 3. Although the obtained pulses must be adapted a little to compensate for the dummy mass(es) during a crash test, this will not be done in this conceptual research because the separate dummy influence is not simulated with the used uncoupled model. To realize the lowest deceleration level of 9 g in the second phase of the optimal deceleration curves, which is independent of the considered crash velocities, the average constant crumple force of the longitudinals can be used. However, to realize a higher deceleration in the first phase with a stiffer structure in the front part is more difficult, because normally deformation starts in the weakest part of the loaded structure. In addition, the desire to obtain different deceleration levels for different collision speeds means that an adaptive structure is needed to adapt the stiffness at the beginning of the crash. Note that in case of adaptable structure stiffness occupant mass corrections are easier to realize by determining the additional vehicle load, and based on this value the structure stiffness must be increased. For the last phase, it is easy to use a stiffer cross-section of the longitudinal to increase the deceleration to a velocity independent level. However, it is more likely that after 586 mm deformation the engine and other components will be impacted against the firewall which generates already a much higher deceleration. It is more important to prevent an early increase in the crash force by the engine. In the next sections concept ideas are presented as a solution for the mentioned problems.

CONCEPT IDEAS FOR ADAPTABLE STRUCTURE STIFFNESS

Energy Absorption by Friction

A practical method to absorb kinetic energy is by means of friction. Changing the pressure force on a friction block regulates the energy absorption. The well functioning idea of hydraulic vehicle brakes can be used during a crash on a backwards moving rod mounted inside the free inner space of both straight longitudinals (which must be positioned in such a way that the rods move under the vehicle floor). The application of friction blocks around a square rigid rod can generate the desired additional deceleration forces. In case of a 64 km/h collision the additional deceleration, next to the 9 g generated by the crushing longitudinals, must be 36 g. For this deceleration an axial friction force of 388 kN is needed in the case of

a 1100 kg vehicle. The choice for friction material must be tungsten, because a high temperature could be expected and melting of the material must be avoided for a necessary high friction coefficient. As calculated in Witteman 1999a, the temperature at the contact surface of the friction block rises 2328 K and on the rod the temperature rises 1698 K, but the temperatures drop very fast inside the material. Therefore both surfaces must contain a coating or plate of tungsten. In this case the expected friction coefficient is at least about 0.45 (normally for different dry metals without lubricator) but in case of two equal metals the friction coefficient could rise to 1.0 or more (Landheer 1993). This means that for a safe value of 0.45, a total normal force of at most 862 kN is needed. For this high force level, a hydraulic system is the right choice. If an available hydraulic pressure of maximal 1350 bar is supposed (the same pressure as in common rail diesel engines), the necessary piston area behind the friction surface must be 6385 mm². For this surface, 10 pistons with a diameter of 30 mm (total 7069 mm²) are sufficient to compensate also some pressure loss in the pipe between a radial piston pump and the pistons. The pistons could be positioned in two rows of five pistons with 20 mm space between, opposite to each other and connected with a frame under the vehicle floor. Of course a strong connection with the car structure is required.

In the case of an offset or an oblique collision where only one longitudinal is directly loaded, it is better to use the additional friction force only on the directly loaded longitudinal. For this reason, two sensors are required inside the bumper, in front of the longitudinal, which detect a contact with an object by means of a pressure force or with radar detection. If only one signal is detected (offset collision), only on the rod in the longitudinal at that side the maximal needed additional friction force must be generated. In the case of two signals (full overlap collision), both rods must be loaded with half of the total needed additional force.

To determine the necessary additional force, the velocity information of the vehicle must be used. Since most modern cars use ABS which continuously detects the speed of each wheel, the current speed (or before the last 100 ms from memory to prevent crash influence) of the car is always well known. With this information the pressure generated by the radial piston pump could be controlled. In case of velocities below 32 km/h it could be zero, in case of velocities up to 64 km/h it must be increased to for example 1350 bar. A radial piston pump with zero regulation (no power loss) can be equipped with electronic

pressure control. Another possibility is to keep the highest pressure always available and control the magnetic valves on each piston (comparable with common rail diesel engines). Probably this is for faster adjustments in the very short time preferable. In Table 4, the required number of opened valves is mentioned. In case of a symmetric collision the number is valid for each longitudinal, in case of an asymmetric collision the number is only valid for the directly loaded longitudinal, the valves of the other longitudinal must be closed. Of course for other collision speeds between 32 km/h and 64 km/h a number between the mentioned numbers could be chosen.

Table 4.
Example of number of opened valves to reach enough pressure for additional friction force

Crash velocity	32 km/h	56 km/h	64 km/h
Symmetric collision	0 valves	3 valves	5 valves
Asymmetric collision	0 valves	6 valves	10 valves

After 179 mm deformation, the additional friction force must be removed. This can be done by moving rapidly the zero regulation rod, which controls the eccentricity of the radial piston pump axle, so the oil flows in the opposite direction back and lifts the pistons from the rod. The movement of the zero regulation rod can be done electronically by the pressure control module or it can be done mechanically by a mechanism connected with the zero regulation rod and activated at the right moment by the crossing rod. Another possibility is the use of a large electronic valve in the common pressure pipe, which releases the pressure rapidly.

The third phase of the deceleration curve starts always at 586 mm deformation length. From this point, the deceleration must be increased from 9 to 23 g as long as the crash lasts. If the engine is involved before this point, which is plausible in smaller cars, a solution could be flexible engine mounting points. In addition, the connection of the engine with the drive line must be movable to prevent high translation forces on the engine too early. All the aggregates must be positioned in such a way that only after 586 mm deformation has been reached, high contact forces start to press the aggregates together. In addition, the front wheels have to deform the wheel bay and the sill. Finally, the engine hits the stiff firewall, which could deform at high collision speeds. The final deformation forces are very dependent on

the positioning and the dimensions of the aggregates and the free space under the bonnet. If the necessary force level is not reached, assistance of the friction force as used in the first phase of the collision is always possible. Signals must be sent to the pressure regulation module and the valves to control the correct friction force.

Future Possibilities

An optimal regulation for the whole deformation length is of course with a computer controlled system, which measures continuously the actual deceleration level and adjusts at the same time the pressure to reach the programmed optimal deceleration pulse. Maybe when very fast sensors, high-pressure valves and control modules are available this is a realizable idea. In this way, it is also possible to compensate for the stiffness, velocity or weight of the colliding obstacle. This would be an ideal solution for the compatibility problem between small and large vehicles. If this system is fast enough and very reliable, it is possible to think about a structure which has only two very stiff beams, which can fully slide backwards without deformation. A heavy computer controlled break system regulates the desired deceleration. Sensors already send signals to increase the friction of one loaded beam to reach the same energy absorption in case of an offset collision. The new beams have not to crumple to absorb energy so they can be made very stiff with a high bending resistance yielding no risk for a premature bending collapse in case of an oblique crash direction. Of course the control system with the breaks must be reliable in all crash situations because there is no alternative to moderate the energy absorption, which means that large force level differences must be taken care of. Only problem could be the space behind the firewall or under the vehicle floor. Vehicles with structural space under the passenger compartment have very good possibilities for safety increasing features, also for side impact crashworthiness. A very nice vehicle concept for this application is the Mercedes-Benz A-class vehicle. Because of the double floor with a higher placed passenger compartment, the longitudinals stay fully horizontally in a stiff ladder chassis. In the floor structure there is enough space for rearward sliding beams and for the positioning of the energy absorbing brake system. Furthermore, the engine does not shorten the available deformation length or penetrate the firewall since it moves to the road surface. Maybe with the popularity of space wagons or mini multi purpose vehicles nowadays, this is an interesting design aspect. Occupants are not longer sitting in the extension of the crumple zone but above, especially at

side impact crashes. In the case of structural space behind the firewall, the hydraulic piston solution presented in next section is another possibility.

Design of a Hydraulically Controlled Frontal Car Structure

To load the missed longitudinal member during an asymmetric collision, it is possible to use a hydraulic system. In Figure 5, a principle sketch of the system is shown with besides the longitudinals two cylinders with pistons.

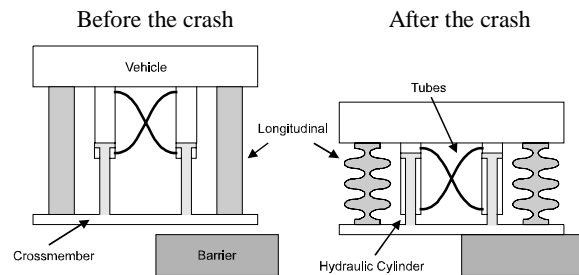


Figure 5. A hydraulically controlled frontal car structure.

The cylinder rods are fixed to the cross member, just like the front ends of the longitudinals. If one of the longitudinals is loaded during an offset crash, it starts to deform and because of the connection to the cylinder, the rod slides into the cylinder. The oil inside the cylinder is pressed via a tube or pipe to the rod side of the cylinder of the unloaded longitudinal. Under the influence of this oil pressure, the piston of this cylinder is also pushed backwards. Because this piston is connected to the unloaded longitudinal member, it is forced to collapse in an axial folding mode. The pressure that arises in the cylinder of the unloaded longitudinal is led back to the rod side of the cylinder of the loaded longitudinal, where it helps to further move the piston inside the cylinder. Hence, the hydraulic cylinders form a closed-loop system. Note that in the case of a full overlap collision where both longitudinals are loaded, the system is in equilibrium and does not influence the crash behavior.

One problem is however, that the oil volume in the cylinder does not fit in the other cylinder at the rod side, because of the volume of the rod itself. Because the rods move inwards, the total available volume decreases. Solution is a piston with at each side a rod, where the second rod has not a force function but causes identical volumes exchanges. For this solution there must be space behind the firewall where the additional rods can move backwards. Advantage is

the same area at each piston side, which gives a 1:1 force transmission.

A second problem is the available deformation length, because a cylinder with piston can be shortened less as half of the original length. For this reason it is also necessary that there is much horizontal space under the passenger floor, because then the cylinders could be mounted at the rear of the firewall. For the connection pipes, enough space is also important because they must have a large diameter and a short length to minimize the pressure loss at high stream velocities. With a cylinder diameter of 90 mm, the pressure at a crash load of 150 kN is 236 bar. At an initial flow rate of 15 m/s (56 km/h crash), the pressure loss in the connection pipe with a diameter of 30 mm and with oil ISO VG2 is 12 bar/m and 1 bar/m for a pipe diameter of 60 mm (Slaats 1996b). Although the guaranteed maximal velocity for the cylinder's sealing is much lower, the high velocity works a very short time and the system has to work only one movement.

The final structure can be built together, the rod of the cylinder can be positioned inside the crushing longitudinal, and gives additional bending resistance. The stiff cylinder behind the longitudinal can be used as support structure for the axial crushing forces.

This hydraulic supported structure generates a constant deceleration force, independent of the overlap percentage. However, to reach the optimal crash pulse, control of the oil flow is necessary. In this case, a valve with a controllable flow restriction (Janssen 1994) or several valves must be used in the outlet of the backside of the cylinders. Reducing the outlet area increases the pressure and therefore the stiffness of the system. After the first deceleration interval, the valve can be fully opened and for the third interval, if necessary the total outlet area can be reduced again.

CONCLUSIONS

A method has been described how a deceleration pulse can be optimized. As an example three pulses are mentioned for three different velocities to use as specification for conceptual design ideas. To fulfil the requirements for different velocities an intelligent structure is desirable. With the use of an additional friction force on rearwards moving rods mounted inside the free inner space of the longitudinals, it must be possible with a hydraulic system to control the deceleration pulse to the optimal level dependent on the crash velocity. In case of a multi purpose vehicle concept (component space under the passenger floor)

a new hydraulic brake or flow system for controlled energy absorption is a promising idea. This intelligent structure with adaptable stiffness is also a solution for the compatibility problem between different vehicles or for compensating the additional occupant and luggage masses.

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